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# SIMULATION OF OVERHEAD CRANE DYNAMICS SUBJECTED TO FRAMEWORK ELASTICITY

*The paper presents an overhead crane dynamics analysis while taking into account its framework elasticity. The elasticity distribution along the framework length being dependent upon trolley position introduces sufficient nonlinearities into the plant model that largely vary the natural vibration frequencies. It is shown that dynamical behavior of an overhead crane can be approximated using a three-mass system, parameters of which are manually or automatically adjusted so that its step response coincides with the distributed parameter model.*

***Keywords: overhead crane, framework vibrations, Comsol Multiphysics, finite-element method, multimass system, parameter estimation, Simulink.***

The payload transportation via different kinds of cranes defines additional specifications for control algorithms. During the transportation the payload sway is induced thus slowing down the entire production process, since the payload has to be brought to rest before it may be lowered. For this problem there are already a set of possible solutions [1, 2] that include optimal and suboptimal trajectory planning, active anti-sway control using the swing angle tracking or estimation etc.

However, as the dimensionality of the crane framework increases it can no longer be treated as rigid, since elastic displacements may be of significant value, which induces undesirable additional mechanical loads leading to reducing the crane lifespan due to possible occurrence of metal fatigue. Since current computational capacities may be implemented as compact device and considering the advancements in measuring technology these vibrations can and should be suppressed, which requires an accurate mathematical model. Most of published works consider framework elasticity either subject to loads due to gravity or as a static analysis. Therefore the presented investigation is considered to be state of the art.

**The aim of this paper** is to develop a mathematical model to represent elastic overhead crane dynamics.

Since the crane framework cannot be treated as rigid object an analysis of its dynamic behavior has to be carried out taking into account its elastic properties. In such a case it is said to be a distributed parameter system. A typical approach to analyze its dynamics is its partition into a set of smaller elastically coupled objects also known as finite elements. The smaller are the element dimensions the higher is the computational accuracy. Thus having partitioned the investigated system into a sufficient number of objects a required precision may be achieved.

Manually building a mathematical model for such a system is very complicated since each of the elements has to be treated as a separate body. This involves computing the stiffness of elastic coupling between every set of adjacent elements. The finite element method based software however allows creating such models quite easyly without having to evaluate stiffness factors or any other framework parameters. Only the information on framework form, its material and available degrees of freedom or movement constraints is required. In this paper a finite element model is constructed using Comsol Multiphysics simulation software, as shown in Fig.1.

This model was constructed taking into consideration several simplifications due to software limitations:

1. The entire construction is modeled as solid body combination, since both "beam" and "shell"-type objects are not suitable for the current application due to several limitation placed within the computational environment. In the meantime modeling a thin solid framework leads to a severe increase in simulation time and memory requirements. Full solid body model however has much higher framework mass. To avoid this material properties are adjusted so that the mass does not exceed real crane parameters without changing obtained natural eigenfrequencies.

2. The trolley is fixed in a given place and is considered to be part of the framework, i.e. no trolley motion is possible.

3. The payload was removed from the model due to the difficulty of implementing the influence of its swing on the crane framework.

With these assumptions this model gives sufficiently accurate information on dynamic behavior of a real crane framework and thus can be accepted as a basis for further investigations and analyses.

The following step in current analysis is the estimation of framework eigenfrequencies, which represent possible displacement modes of the crane framework subject to different loads. This can also be performed automatically using Comsol Multiphysics simulation software. The first four eigenmodes and corresponding frequencies are shown in Fig.2.

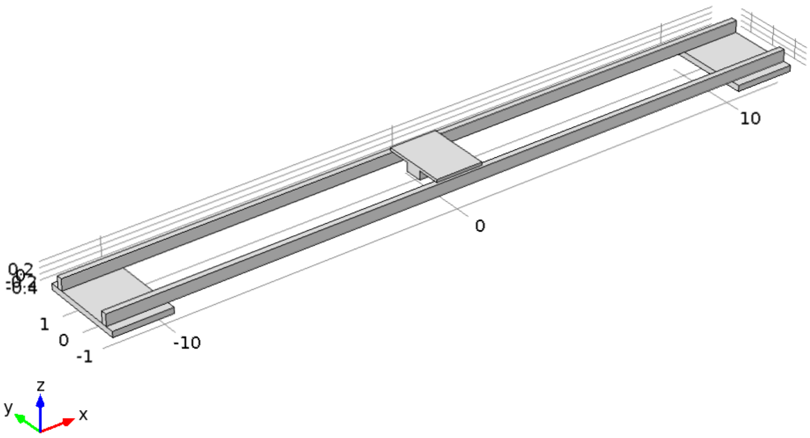
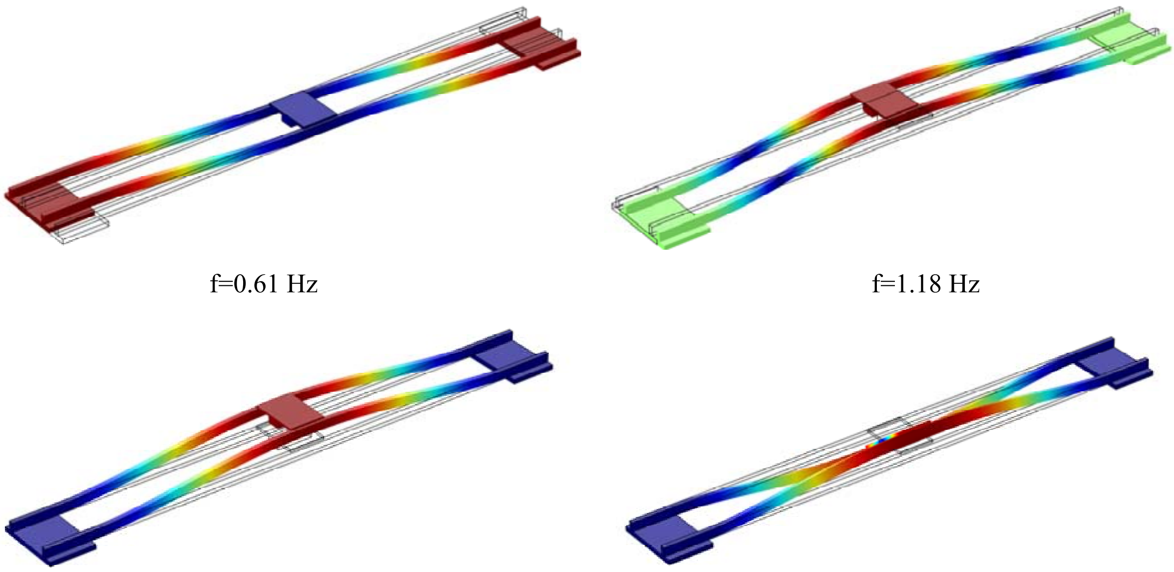



*Figure 1 – Finite element model of crane framework*



f=0.61 Hz f=1.18 Hz

f=1.76 Hz f=3.31 Hz

*Figure 2 – First four framework eigenmodes*

The applied color map depicts the total displacement distribution along the framework with red marking the larges displacement, and blue – the lowest. Since trolley motion in this model is constrained and considering an ordinary overhead crane construction, the only way to suppress the framework swing is through modification of bridge drive control algorithm. Therefore, only framework vibration due to bridge motion, i.e. along Y-axis, is considered.

The frequency spectrum of a real crane framework however may contain much more than these components. Therefore, a transient response has to be obtained to define the most substantial frequency components.

The model is now subjected to a bang-bang force applied to its outer boundaries with equal distribution. Although displacement signal can be measured, it is more reasonable to measure velocities, because it would contain much more information on present vibration frequencies than a displacement signal and at the same time much less noise than acceleration signal. The results in Fig.3a present velocity time histories, measured in three points of the framework: $v_1$ and $v_3$ – on the outer boundaries, $v_2$ – on a trolley.

It is clear that there is only one substancial eigenfrequency present in this signal, which coincides with shown in Fig.2 eigenmode of 1.18 Hz. For this case a state-feedback controller was designed in [3]. Presented results indicate efficient framework vibration suppression. However, those results were obtained under condition that the trolley is fixed precisely in the framework center. When this is not the case, the velocity time histories may vary greatly from these results as shown in Fig.3b. In this case the trolley was set at 5 and 15 m respectively from the outer boundaries.

It is clear that the framework eigenfrequencies have changed considerably. The proposed state-feedback controller is known to be highly susceptive to plant parameter change end estimation accuracy. Therefore, it can be concluded that a simple state-feedback controller cannot handle such task.

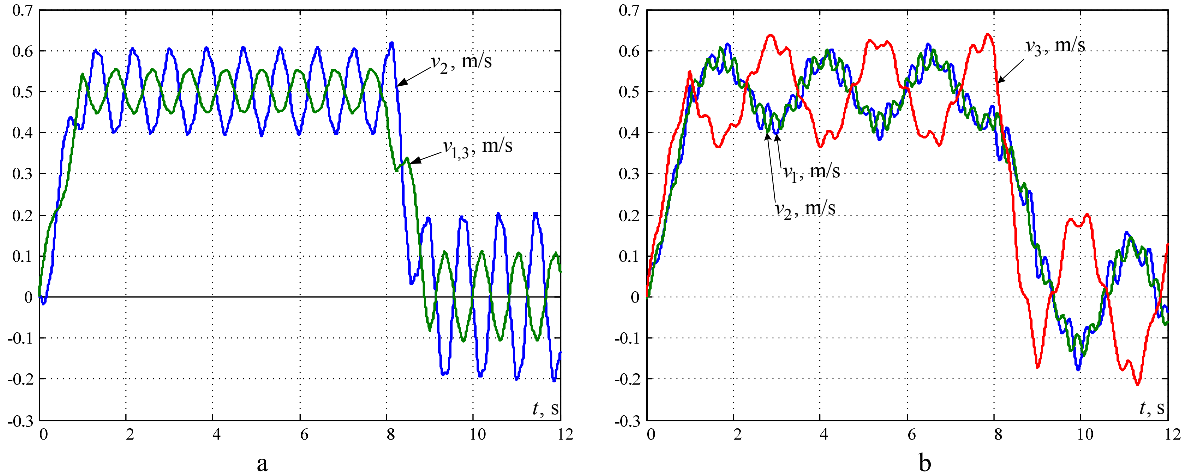



*Figure 3 – Velocity time histories with trolley positioned in the middle (a) and closer to one end (b)*

The described plant has significant nonlinearities that influence greatly its dynamical behavior. In a first approximation such plant can be reduced to a lumped parameter system, i.e. multi-mass system. In this case the number of lumped masses is determined by the number of present harmonics in plant time response, which is in our case equals two. Therefore, a three-mass system is considered which can be described with following equation of motion:

$$\begin{cases} m_1 \dfrac{dv_1}{dt} = F_1 - c_{12}(s_1 - s_2), \\ m_2 \dfrac{dv_2}{dt} = c_{12}(s_1 - s_2) + c_{23}(s_3 - s_2), \\ m_3 \dfrac{dv_3}{dt} = F_3 - c_{23}(s_3 - s_2), \\ \dfrac{ds_1}{dt} = v_1, \ \dfrac{ds_3}{dt} = v_3, \\ \dfrac{ds_2}{dt} = v_2, \end{cases} \tag{1}$$

where $m_1$, $s_1$, $v_1$ and $F_1$ – equivalent mass, position, velocity and applied force of the first body; $m_3$, $s_3$, $v_3$ and $F_3$ – equivalent mass, position, velocity and applied force of the third body; $m_2$, $s_2$, $v_2$ – equivalent mass, position and velocity of the second body, located between the first and the third ones; $c_{12}$ and $c_{23}$ – stiffness factors of elastic couplings between the bodies.

If the trolley is located in the middle point of the crane, its model can be easily reduced to a three-mass system. The lumped masses and stiffness can be calculated as shown in [3, 4]. However, there are no exact formulas to estimate these parameters in any other case, because the crane framework can no longer be treated as fixed beam subjected to point load. If the trolley is not located in the middle point of the crane, framework outer points, i.e. crane wheels, no longer move synchronously. There is a position difference between them, which is not taken into account when reducing a framework dynamics to beam vibrations.

However, the required parameters can be evaluated approximately, i.e. with numerical approximation methods. Considering the information from Fig.3 it can be seen that there are at least two natural frequencies with significant amplitudes that should be contained in a model. That leaves us at least four parameters to adjust two stiffness factors as well as mass ratios. Fig.4 shows the simulation results of finite-element model and three-mass system, created as a block diagram in Simulink simulation environment for two different trolley position, defined with respect to the middle point. It is obvious, that this model is a bit inaccurate. The farther the distance from the trolley position to middle point of the framework, the less accurate becomes the model, which indicates, that such model reduction is suitable for control algorithm design but requires additional refinements relative to its parameters. Future works should be focused on designing and testing accurate control techniques using such a model.

**Conclusions:** 1. Finite-element analysis of dynamical behavior of elastic crane framework gives information on vibration frequencies and amplitudes, which can be further used to create simple models.

2. Trolley position introduces nonlinearity into the plant model so that it changes its properties in a wide range.

3. Dynamics of elastic overhead crane framework may be approximated by three-mass system, parameters of which should be estimated numerically. Such model depicts desired system behavior and can be used to design control techniques.

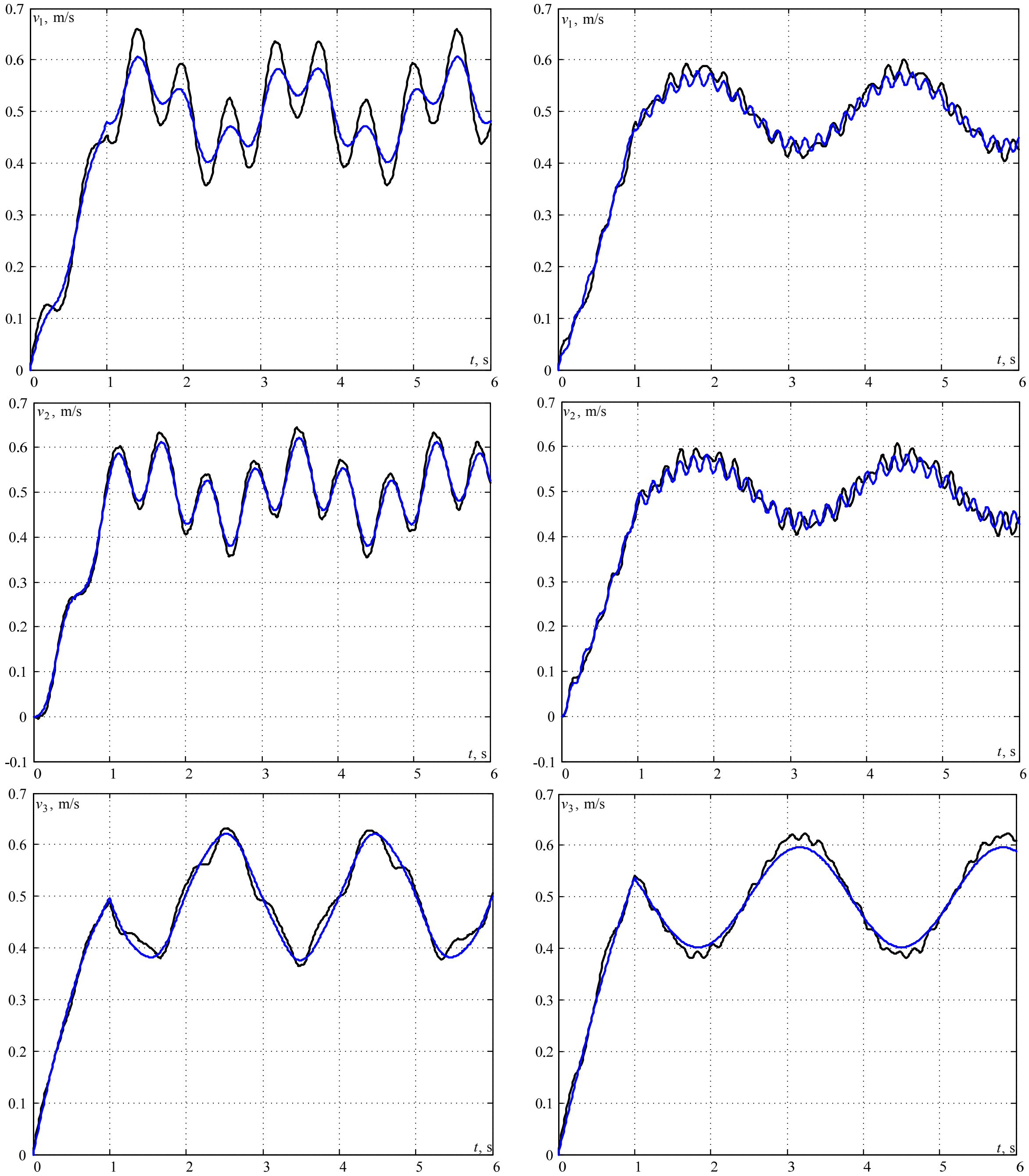



*Figure 4 – Velocity time histories from finite element (black) and Simulink (blue) models: left – trolley position -3 m; right – trolley position -6m*

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**Моделювання динаміки мостового крану із урахуванням пружності конструкції.** Наведено аналіз динаміки мостового крану з урахуванням пружних властивостей його конструкції. Розподіл пружності уздовж конструкції мосту залежить від положення візка на ньому і зумовлює наявність нелінійності у моделі об'єкта у вигляді зміни частот власних коливань. Показано, що динаміку мостового крану можна приблизно описати за допомогою тримасової системи, параметри якої необхідно налаштовувати вручну або автоматично так, щоб її результати її роботи співпадали з моделлю з розподіленими параметрами.
***Ключові слова: мостовий кран, коливання конструкції, Comsol Multiphysics, метод кінцевих елементів, багатомасова система, визначення параметрів, Simulink.***

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**Моделирование динамики мостового крана с учетом упругости конструкции.** Приведен анализ динамики мостового крана с учетом упругих свойств его конструкции. Распределение упругости вдоль конструкции моста зависит от положения тележки на нем и обуславливает наличие нелинейности в модели объекта в виде изменения частот собственных колебаний. Показано, что динамику мостового крана можно приблизительно описать с помощью трехмассовой системы, параметры которой подстраиваются вручную или автоматически так, чтобы результаты ее работы совпадали с моделью с распределенными параметрами
***Ключевые слова: мостовой кран, колебания конструкции, Comsol Multiphysics, метод конечных элементов, многомассовая система, определение параметров, Simulink.***